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Continuous vs. pulsating flow boiling. Part 2: Statistical comparison using response surface methodology

Martin Ryhl Kærn[1]*, Brian Elmegaard[1], Knud Erik Meyer[1], Björn Palm[2], Jørgen Holst[3]

[1]Technical University of Denmark, Department of Mechanical Engineering, Kongens Lyngby, Denmark,
Tel: +45 4525 4121, Fax: +45 4593 5215, Email: pmak@mek.dtu.dk

[2]Royal Institute of Technology, Department of Energy Technology, Stockholm, Sweden

[3]Danfoss Drives A/S, Gråsten, Denmark

ABSTRACT

Response surface methodology is used to investigate an active method for flow boiling heat transfer enhancement by means of fluid flow pulsation. The flow pulsations are introduced by a flow modulating expansion device and compared with the baseline continuous flow provided by a stepper-motor expansion valve. Two experimental designs (data point sets) are generated using a modified Central Composite Design for each valve and their response surfaces are compared using the quadratic model. Statistical information on the significant model terms are used to clarify whether the effect of fluid flow pulsations is statistically significant in terms of the time-averaged flow boiling heat transfer coefficient. The cycle time range from 1 s to 9 s for the pulsations. The results show that the effect of fluid flow pulsations is statistically significant, disregarding the lowest heat flux measurements. The response surface comparison reveals that the flow pulsations improves the time-averaged heat transfer coefficient by as much as 10 % at the smallest cycle time compared with continuous flow. On the other hand, at highest cycle time and heat flux, the reduction may be as much as 20 % due to significant dry-out when the valve is closed. These values are higher than reported in part 1 of the paper, but evaluated more consistently at equal heat flux using the response surfaces.

1. INTRODUCTION

Design of Experiments (DOE) is a method for experimentation that enables the possibility to estimate effects and responses of certain factors and their interactions with statistical significance and low experimental effort. This approach includes several phases from sorting out the trivial from the vital factors, characterizing main effects, interactions, detecting non-linearity and establishing functional relationships (Response Surface Methodology, RSM) for optimizing and comparing responses etc. In the field of heat and mass transfer (or exchangers), there has been numerous studies aimed on creating simplified performance models using RSM for optimization purposes. For example, Pandelidis and Anisimov (2016) developed simple performance models of the cross-flow Maisotsenko cycle heat exchanger. Salviano *et al.* (2015) used a Computational Fluid Dynamics (CFD) model and RSM to optimize vortex generator positions and angles in a plate-fin compact heat exchanger. Han *et al.* (2014) employed CFD and RSM to optimize various design parameters of outward convex corrugated tubes.

The motivation of using response surface methodology in the current experimental investigation builds upon the relatively small heat transfer improvements that were shown in part 1 of the paper for pulsating in-tube flow boiling relative to continuous in-tube flow boiling. The improvements were on average 3.2 % in the time-averaged heat transfer coefficient at low cycle time (1 s to 2 s), whereas a reduction by as much as 8 % were found at high heat flux ($q \geq 35$ kW/m^2) and cycle time. At the same time, the experimental error of the predicted heat transfer coefficients were found to be 3.8 % by using the error propagation method by Kline and McClintock (1953) for most of the measurements ($\alpha \geq 2$ kW/m^{2}K). Accordingly, the improvements were similar to the experimental error and thus questionable. Response surface methodology provides statistical information that may be used to answer this question, i.e. whether the improvements are statistically significant (verification of the pulsation effect).

Another benefit of the response surface methodology is the establishment of functional relationships. These functional relationships are empirical models or fits of the experimental data; however, they may be used to compare the current experiments more consistently. In typical experimental setups that are used to measure in-tube continuous flow boiling heat transfer coefficients, both the vapor quality and the heat flux are controllable. This is not possible in the current setup, because the state before the expansion valve, which provides either continuous (stepper-motor valve) or pulsating flow (flow modulating valve) to the evaporator test section, must be maintained equal to ensure a fair comparison. It results in a fixed inlet vapor quality to the test section. Thus, variation in the two-phase heat transfer coefficient will result in different local vapor quality and heat flux for the same boundary conditions (inlet fluid states and flows) in the current co-axial evaporator test section. Even if electrical heat was used and the heat flux were maintained the same, the vapor quality would still differ. Moreover, the current heat transfer coefficients were compared directly in their exact locations in the evaporator test section in part 1 of the paper, for which the above-mentioned improvement/reduction (3.2 % and 8 %) adheres to. This comparison was influenced by differences in both vapor quality and heat flux, while the refrigerant mass flux and water volume flow were held constant. Using the functional relationships (response surfaces), the heat transfer coefficients may be compared at even vapor quality and heat flux too.

The objective of the current paper is to verify the effects of fluid flow pulsation on the time-averaged flow boiling heat transfer coefficient that were found in part 1 of the paper, and determine whether the effects are statistically significant using response surface methodology. Additionally, the investigation brings a deeper and more consistent comparison of the continuous and pulsating in-tube flow boiling.

2. EXPERIMENTAL APPARATUS

The experimental apparatus has been presented in part 1 of this paper. The experiments were performed with two exchangeable expansion valves, namely the flow modulating expansion valve (Danfoss AKV) and the stepper-motor expansion valve (Danfoss ETS), i.e. pulsating and continuous flow, respectively. An oil-less refrigerant pump loop was used to create standard conditions before (T_{sat} = 32°C, T = 30°C) and after (T = 5°C) the expansion valves. Moreover, the states before the expansion valves and the evaporation temperature were kept constant in all the experiments conducted. The test section was placed immediately downstream of the expansion valve with only a glass section in between for visualization. The test section was a co-axial tube evaporator with R134a flowing internally and distilled water flowing externally. It consisted of four sub-sections each capable of measuring the heat transfer coefficient. Moreover, the water temperatures in and out of each sub-section, the wall temperatures in the center of each sub-section and the temperature and pressure of the refrigerant were measured. The inner tube was made from copper and had an internal diameter of 8 mm. Furthermore, the inlet water temperature was held constant at 25°C. The water volume flow and refrigerant mass flux were the only factors varied for the continuous flow experiments, whereas the cycle time was a factor for the pulsating flow experiments too.

2.1 Data reduction

The data reduction method, detailed in part 1 of the paper, is similar to the method used by Wojtan *et al*. (2005) and illustrated in Figure 1. The figure shows the enthalpy profiles that were reduced from a single experimental run, for which the enthalpy reference were changed to zero at both fluid outlets, respectively.

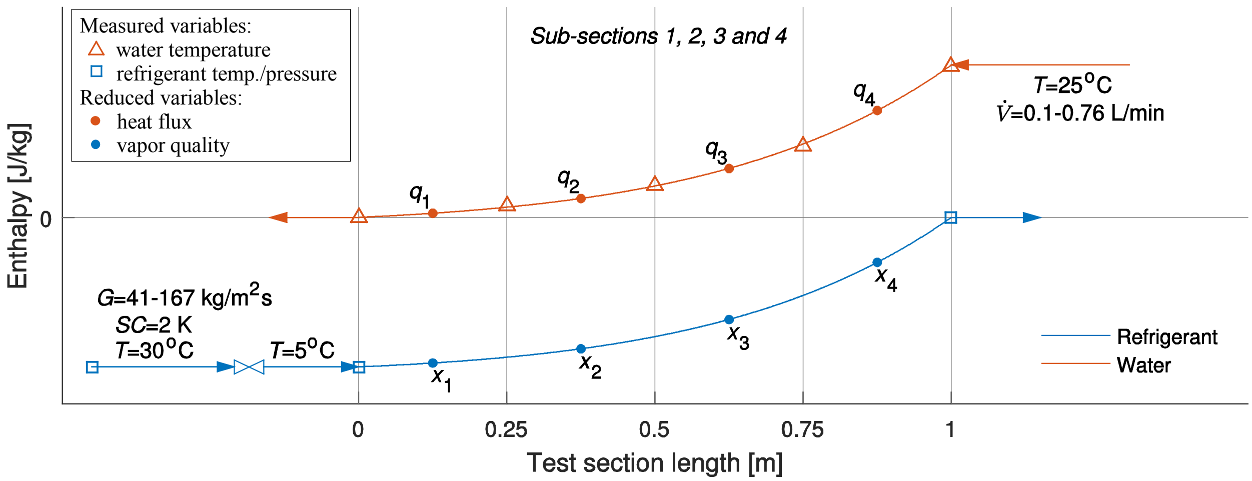

Figure 1: Experimental boundary conditions and reduced enthalpy profiles from a single experiment (the enthalpy reference is changed to zero at each fluid outlet).

The data reduction method relies on the establishment of the water enthalpy profile from the water temperature readings. The curve is differentiated to calculate the local heat fluxes (q_1 to q_4) and changed by the ratio of mass flows to calculate the refrigerant enthalpy profile and the corresponding vapor qualities (x_1 to x_4). It means that the heat flux and vapor quality are correlated and are impossible to vary independently in the current design. Moreover, x and q are outputs from each experimental run and dependent on the heat transfer performance of the continuous and pulsating flow boiling. In other words, creating a response surface in terms of both x and q is meaningless using the current design and more test sections in series are needed to establish such a surface.

Note that all measured variables are time-averaged before the data reduction is conducted. The calculation of the heat flux and heat transfer coefficient is shown in part 1 of this paper. Furthermore, the heat transfer coefficients from sub-section 1 were disregarded because one of the thermocouples, soldered in grooves in the wall, were broken during assembly.

2.2 Experimental design

For in-tube continuous flow boiling in traditional tubing, the governing vital factors have been known for decades and heat transfer correlations typically employ the following factors to correlate the heat transfer coefficient

$$\alpha = f(G, q, x, T, d, \text{fluid}) \tag{1}$$

where G, q, x, T, d are mass flux, heat flux, vapor quality, saturation temperature and diameter, respectively. As already mentioned, the heat flux and vapor quality were impossible to vary independently, and thus the water volume flow and the refrigerant mass flux were chosen as factors for the experimental design (data points measured). For pulsating flow boiling, the injection frequency (or cycle time) was also a factor. In fact, the water volume flow and refrigerant mass flux are correlated too by the energy equation (disregarding heat losses)

$$\dot{V}_w \rho_w c_{p,w} \Delta T_w = G A \Delta h \tag{2}$$

where $\dot{V}_w$, ρ_w, and $c_{p,w}$ are the volume flow rate, density and specific heat capacity of water, respectively, and A and h are the internal tube cross-sectional area and refrigerant enthalpy, respectively. Figure 2b shows the dependency for full evaporation when the water temperature difference (ΔT_w) is fixed at 15 K. We strived to cover as much of the region before full evaporation in our experimental design as indicated in Figure 2b. Figure 2a shows both the continuous and the pulsating flow experimental designs, where the continuous flow points are located hypothetically at zero cycle time.

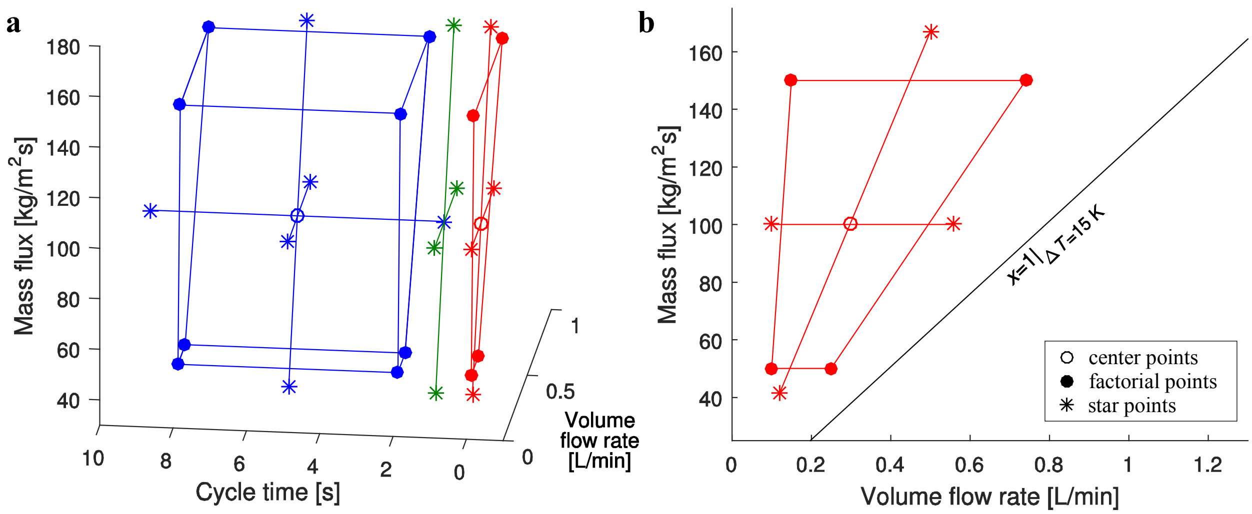

Figure 2: Full experimental design (a) and continuous flow design (b).

The experimental design is a stretched Central Composite Design (CCD) in the non-orthogonal factors, i.e. the refrigerant mass flux and the water volume flow. Five center points were used for both the 2-dimensional continuous flow design and the 3-dimensional pulsating flow design in order to estimate lack of fit in the responses. Additional star points were added at the smallest cycle time (1 s) for the pulsating flow design, since the best performance were believed here by intuition. Furthermore, the factorial points at 8 s cycle time were repeated once to reduce the leverage at these points. The star points were located at an α-value of 1.33, which is close to the recommendation by Whitcomb and Anderson (2005), which is the fourth root of the number of factors $\sqrt[4]{3}$. We generated other experimental designs too using Face Centered Design (FCD), optimal IV and A, however, the standard error in the design for the full quadratic model gave similar results. We used the software Design Expert 8 (2010) to generate and analyze these designs, and to perform the response surfaces and Analysis of Variance (ANOVA) etc. as will be shown in Section 3.

3. RESULTS

This section presents the results of the statistical analysis and the response surfaces with confidence intervals. Although the experimental design was constructed using the volume flow of water as a factor, the heat flux will be used instead, since the heat flux is one of the main factors for flow boiling as indicated by Eqn. 1. The heat flux is chosen over vapor quality, because the experiments indicated nucleate boiling dominance as shown in part 1 of the paper. Moreover, the exponential relationship between the heat transfer coefficient and the heat flux was shown to agree with that proposed for nucleate boiling in the VDI Heat Atlas (Verein Deutscher Ingenieure, 2010), indicating nucleate boiling dominance without significant dependency on mass flux or vapor quality. The experimental design in terms of heat flux and mass flux is shown in Figure 3a for the continuous flow experiments and may be compared with that in Figure 2b from which it was developed. The mass flux range from (41 to 167) kg/m^{2}s, vapor quality from 0.18 to 0.59 and heat flux from (1.5 to 45) kW/m^2. Furthermore, the interdependency of heat flux and vapor quality is shown in Figure 3b, for which a second order polynomial fit is shown for each refrigerant mass flux results presented.

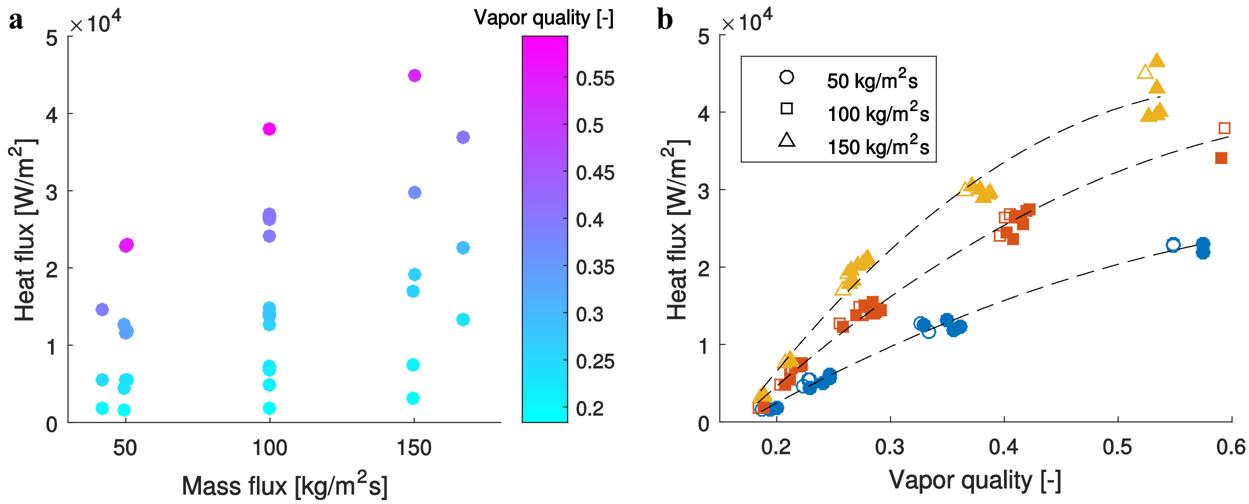

Figure 3: Resulting experimental design in terms of heat flux and refrigerant mass flux for continuous flow (a). Heat flux and vapor quality dependency at different refrigerant mass flux (b), open markers = continuous flow; full markers = pulsating flow experiments)

3.1 Response and ANOVA

Two response surfaces were generated using the full quadratic model for both the continuous and the pulsating flow. The responses were transformed using the Box-Cox plot power transform recommendation and the resulting response surfaces became

$$\begin{aligned} \alpha_{\mathrm{cont}}{}^{1.23} = {} & +2103.0 + 3.9714\,G + 0.69413\,q \\ & + 2.6110\text{e-}3\,G\,q \\ & - 2.4341\text{e-}2\,G^2 + 1.7448\text{e-}6\,q^2 \end{aligned} \tag{3}$$

$$\begin{aligned} \alpha_{\mathrm{puls}}{}^{0.77} = {} & +80.509 + 0.67340\,G + 2.3052\text{e-}2\,q + 7.3014\,ct \\ & + 3.4412\text{e-}5\,G\,q - 9.9511\text{e-}3\,G\,ct - 6.9404\text{e-}4\,q\,ct \\ & - 1.4793\text{e-}3\,G^2 - 2.3949\text{e-}7\,q^2 - 0.20181\,ct^2 \end{aligned} \tag{4}$$

Table 1 and 2 shows the analysis of variance (ANOVA) for the continuous and pulsating flow, respectively. These are based on "partial sum of squares – type III" which is the Design Expert default approach to ANOVA, and means that the total sums-of-squares for the model terms does not add up to that of the model sum-of-squares for non-orthogonal data. The total sums of squares are a measure of the variability in the measurements, and is decomposed into each terms in the model equation (the quadratic model) and a residual. The mean sums of squares for the full model and each term are then compared to the mean residual variability (MSE) not explained by the model equation in order to test the null hypotheses, i.e. that there is no variability. The relevant test is the F-test that compares the F-value (MS_i/MSE) to the critical F-value that assumes the null hypotheses and follows an F-distribution with the same degrees of freedom and confidence. If the F-value is larger than the critical F-value the null hypotheses can be rejected and it means that the effect is statistically significant. The probability value (p-value) expresses probability of seeing the observed F-value if the null hypothesis is true. By default, Design Expert indicates significance when the p-value is below 0.05 and insignificance when the p-value is above 0.1. These values may be chosen arbitrarily and indicate confidence, e.g. if the p-value is below 0.05 there is a 95 % confidence of the effect. The same p-values and confidence will be adopted herein.

For the continuous flow response, it means that the second order effects are not significant, while the linear terms and the interaction term are significant. For the pulsating flow response, the interaction term and second order terms for the mass flux and the cycle time (*ct*) are insignificant while the other terms are significant. The linear term for cycle time and the interaction with the heat flux is thus significant in the model. Whether it is the improvement or the reduction in the heat transfer coefficient that is significant in terms of cycle time will be detailed in Section 3.3.

Table 1: ANOVA (continuous flow)

Source	Sum of Squares (SS)	df	Mean Square (MS)	F Value (MS_i)	p-value Prob > F
Model	6.69e+09	5	1.34e+09	2262.2	< 0.0001
A-*G*	1.10e+08	1	1.10e+08	186.2	< 0.0001
B-*q*	2.39e+09	1	2.39e+09	4045.8	< 0.0001
AB	2.49e+07	1	2.49e+07	42.02	< 0.0001
A^2	5.15e+04	1	5.15e+04	0.087	0.7695
B^2	1.41e+06	1	1.41e+06	2.391	0.1301
Residual	2.31e+07	39	5.92e+05		
Lack of Fit	2.22e+07	21	1.06e+06	20.66	< 0.0001
Pure Error	9.19e+05	18	5.11e+04		
Cor Total	6.72e+09	44			

Table 2: ANOVA (pulsating flow)

Source	Sum of Squares (SS)	df	Mean Square (MS)	F Value (MS_i)	p-value Prob > F
Model	2.78e+06	9	3.09e+05	466.1	< 0.0001
A-*G*	5.36e+04	1	5.36e+04	80.81	< 0.0001
B-*q*	5.44e+05	1	5.44e+05	820.6	< 0.0001
C-*ct*	5.99e+04	1	5.99e+04	90.39	< 0.0001
AB	7.19e+03	1	7.19e+03	10.85	0.0015
AC	8.77e+01	1	8.77e+01	0.132	0.7171
BC	2.83e+04	1	2.83e+04	42.72	< 0.0001
A^2	2.79e+02	1	2.79e+02	0.421	0.5186
B^2	5.67e+04	1	5.67e+04	85.53	< 0.0001
C^2	1.08e+02	1	1.08e+02	0.163	0.6875
Residual	4.90e+04	74	6.63e+02		
Lack of Fit	4.86e+04	50	9.72e+02	51.29	< 0.0001
Pure Error	4.55e+02	24	1.89e+01		
Cor Total	2.83e+06	83			

The pure error sum of squares is based on the repeated experimental runs in the experimental design, e.g. the center points plus additional factorial points. These data points show very small discrepancies and result from a good repeatability in the measurements. On the other hand, the lack of fit sum of squares in the models become high relative to the pure error sums of squares and the ANOVA results in a significant lack of fit. Note that the sum of the pure error and lack of fit sum of squares equals the residual sum of squares, i.e. the residual variability that are not modeled by the quadratic model terms. Moreover, because the pure error becomes low, the lack of fit becomes high. Design Expert reports the adjusted R^2 values as 0.996 and 0.981 for the continuous flow and the pulsating flow response surfaces, respectively. The R^2 measures the amount of variability explained by the model equation relative to the total variability, thus indicating good responses. Another measure of the goodness of the fits are the mean averaged deviation (MAD) between the experimental data and the models, and were calculated to be 3.86 % for the continuous flow and 5.67 % for the pulsating flow results, respectively, where the MAD is computed by

$$\mathrm{MAD} = \frac{1}{N}\sum_{i=1}^{N}\left|\frac{x(i)_{\mathrm{pred}} - x(i)_{\mathrm{exp}}}{x(i)_{\mathrm{exp}}}\right| \tag{5}$$

Both the MAD and R^2 value seems sufficient for comparing the response surfaces.

3.2 Comparison of response surfaces

Figure 4a shows the heat transfer responses compared at equal heat fluxes vs. refrigerant mass flux including the confidence intervals. Figure 4b shows the normalized responses based on the continuous flow response.

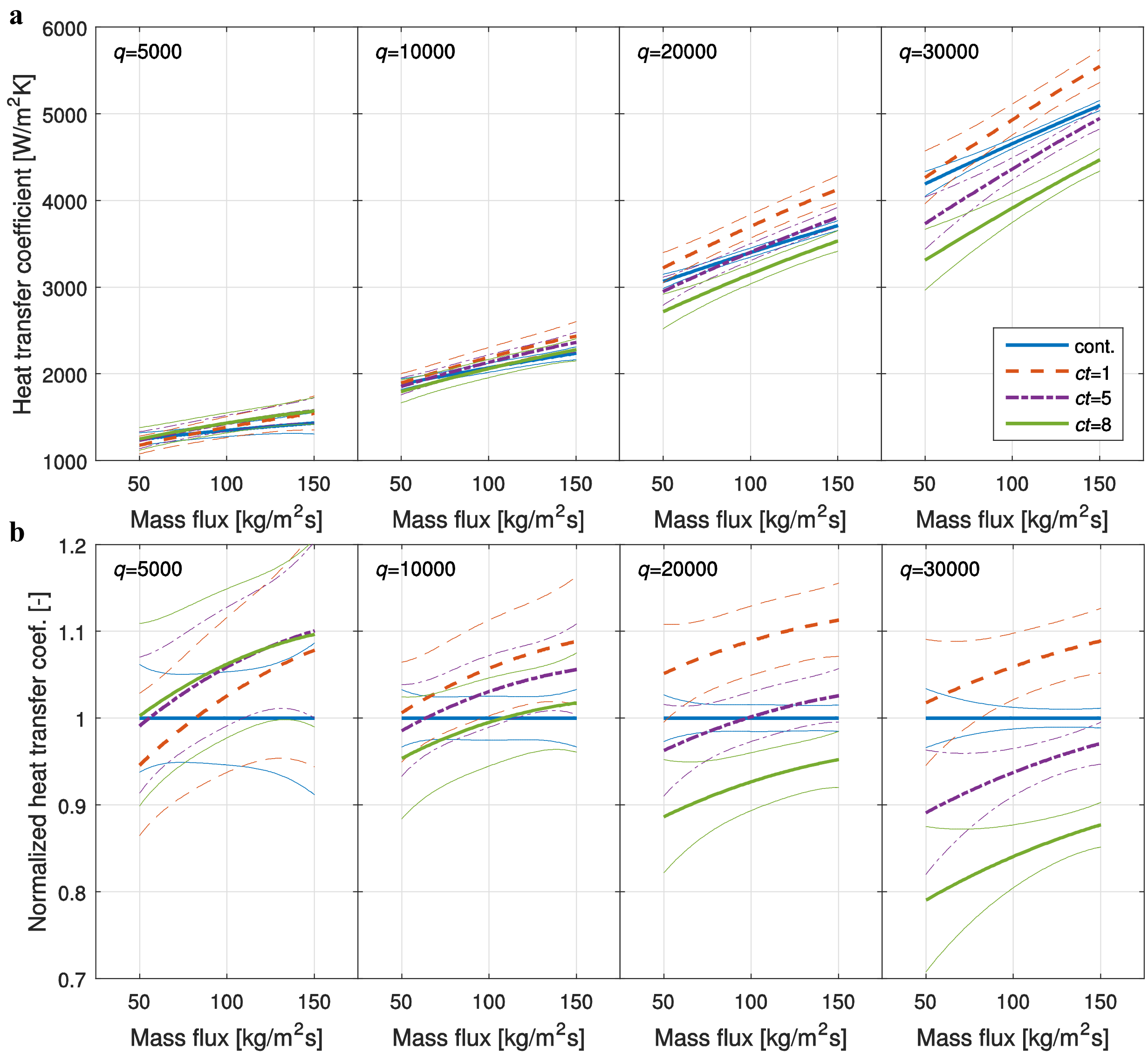

Figure 4: Heat transfer response (a) and normalized heat transfer response based on continuous flow response (b). Thick lines represent the response; thin lines represent confidence intervals.

With overlapping confidence intervals, we would need to accept the null hypotheses (insignificant effect). This is the case for the low heat flux measurements. It does not mean that there is no effect of flow pulsations at low heat flux, only that the effect cannot be verified with these measurements. At higher heat flux and mass flux, the confidence intervals tends towards no overlaps and we may reject the null hypotheses, i.e. the flow pulsations are statistically significant for the improvements at 1 s cycle time disregarding the low heat flux measurements. Similarly, the reductions at high cycle time and heat flux are statistically significant too.

The results are similar to part 1 of this paper as expected; however, the relative differences are larger. The fastest pulsations at a cycle time of 1 s gives the best performance, disregarding the response at low heat flux (q = 5 kW/m^2), and results in as much as 10 % improvement at the highest mass flux. Additionally, there is a tendency towards better heat transfer with pulsations as the mass flux increase. Again, a high heat flux and cycle time leads to poor heat transfer performance (here 10 % to 20 % reduction), since the wall dries out significantly as was shown in part 1 of this paper.

The response at a heat flux of 5 kW/m^2 shows a reduction in the heat transfer coefficient with smaller cycle time. This is not similar to what was found in part 1, where the pulsations were found to increase the heat transfer coefficients at lowest cycle times. We analyzed the data below 10 kW/m^2 independently and found that the reduction in the heat transfer coefficient with cycle time in Figure 4b at q = 5 kW/m^2 is erroneous. Moreover, the curvature as function of cycle time at high heat fluxes seems to propagate through the response surface, and results in a change of the slope with cycle time at low heat fluxes. On the other hand, the actual differences in the curves with flow pulsation are very small as indicated in Figure 4a at q = 5 kW/m^2 and the confidence intervals overlap.

3.3 Statistical significance

The confidence intervals in Figure 4 show where we may accept or reject the null hypotheses. To strengthen the statistical analysis, we created response surfaces for which both the continuous flow results and the individual pulsating flow results (at each cycle time) were analyzed categorically. In this way, the continuous flow results could be analyzed against the flow pulsation results at each cycle time independently of other cycle time results. Table 3 reports the p-values of the linear cycle time term at different cycle times for the full quadratic model without interaction terms with cycle time.

Table 3: Statistical significance of individual cycle times relative to the continuous flow response

	Cycle time			
p-value, (Prob > F)	**1 s**	**2 s**	**5 s**	**8 s**
C-*ct*	0.0008	0.0425	0.5705	0.0003

The results show that the cycle time is indeed significant at the low and high cycle times (p-value << 0.05 at 1 s and 8 s). The cycle time is however insignificant at 5 s (p-value >> 0.1) indicating no significant effect of the flow pulsations relative to the continuous flow results. Again, it shows that the improvements at low cycle time and the reductions at high cycle time and heat flux using flow pulsation are statistically significant. It suggests that the effect of the cycle time and thus the flow pulsations are valid. At a cycle time of 2 s, the p-value shows that the cycle time term is just significant (p-value < 0.05), and indicates a 95 % confidence of the effect on the response.

4. DISCUSSION

The statistical investigation conducted herein was positive. It was found that the effect of cycle time and the pulsations were statistically significant regarding both the improvements and the reductions in the flow boiling heat transfer coefficient, disregarding the lowest heat flux measurements. The experiments conducted herein are the first obtained from the test rig, and it will be expanded in the near future (more test sections) to include the full evaporation region including dry-out. The expanded test rig will enable the possibility to include vapor quality in the statistical analysis. Moreover, the heat flux and vapor quality would not be dependent and may be varied individually in the additional test sections. Adding the vapor quality to the current RSM analysis would create 3-dimensional and 4-dimensional responses for the continuous and pulsating flow, respectively, thereby increasing complexity and broaden the confidence intervals on Figure 4.

For the current results, the vapor quality is integrated into the value of heat flux and small variations may occur at a given heat flux and refrigerant mass flux. This is depicted on Figure 3b, for example at G = 50 kg/m^{2}s and q = 23 kW/m^2, the vapor quality may differ by 0.025. These small variations are considered negligible. As already mentioned, the results of the RSM comparison at equal heat flux give higher percent improvements and reductions in the heat transfer coefficient. These differences are believed to be a result of having equal heat flux in the comparison, and not influenced much by small variations in vapor quality.

It is interesting to note the tendency towards higher improvements by pulsations as the mass flux increases. This is in contrast to what was expected before the tests were performed, i.e. that the low mass flux conditions with nucleate boiling dominance would benefit by the increased convection due to the flow pulsations. Apparently, the results indicate that the improvements are higher when the convective contribution is higher too, i.e. at higher refrigerant mass flux.

Even though the response surfaces could be used to compare the heat transfer coefficients at equal heat flux and vapor quality, it might be better to create actual correlations for each cycle time, when the full evaporation region is considered. Moreover, the effect of subcooling with and without flow pulsations is something not investigated yet as well as other fluids, mixtures, pressures before and after the expansion valve, etc. In the light of the many factors that may show more or less effect on the heat transfer coefficient, the results from this analysis should be viewed as indications, noting that the investigation was adhered to the location immediately downstream the expansion valve.

The possible improvement in the time-averaged heat transfer coefficient (10 % at maximum) was less than expected with flow pulsations. Compared with other flow boiling heat transfer enhancement techniques, it does not seem as promising. For example, micro-fin tubes with numerous (~70) low height (~0.15 mm), spiraling (~18 deg.) fins improve heat transfer coefficient by as much as 50 % to 100 % (Bergles, 2003), however, the combination with flow pulsations might boost the improvements even more.

5. CONCLUSION

This paper presents a comparison and a statistical analysis using response surface methodology of flow boiling heat transfer in an 8 mm round tube with and without fluid flow pulsation for evaporating R134a. The fluid flow pulsations were generated using a flow modulating expansion valve and analyzed against the baseline continuous flow provided by a stepper motor expansion valve. The mass flux ranged from 41 kg/m^{2}s to 167 kg/m^{2}s, vapor quality from 0.18 to 0.59, heat flux from 1.5 kW/m^2 to 45 kW/m^2, and cycle time from 1 s to 9 s in these experiments. The statistical analysis indicated that the flow pulsations are statistically significant at low (1 s to 2 s) and high (8 s to 9 s) cycle times disregarding the lowest heat flux measurements with overlapping confidence intervals. In usual cycle time range (5 s) the effect of fluid flow pulsations are statistically insignificant compared with continuous flow boiling. The response surface comparison reveals that the flow pulsations improve the time-averaged heat transfer coefficient by as much as 10 % at the lowest cycle time compared with continuous flow. On the other hand, at highest cycle time and heat flux, the reduction may be as much as 20 % due to significant dry-out when the valve is closed. These numbers are larger than reported in part 1 of the paper, but evaluated more consistently at equal heat flux using the response surfaces. Furthermore, the results indicate a tendency towards higher improvements by flow pulsations with increased mass flux.

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